

Modeling of micro-CHP (combined heat and power) unit and evaluation of system performance in building application in United States



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ABSTRACT

In this paper, a model of a micro-CHP (combined heat and power) system is developed and validated with laboratory experimental results. The model is tuned to match steady state experimental test results, and validated with transient experimental results. Further simulations were performed using a modeled thermal storage system, and integrating the CHP system into a building model to evaluate the feasibility of the CHP system in the mid-Atlantic region as well as the Great Lakes region. The transient simulation outputs are within 4.8% of experimental results for identical load profiles during a simulated summer week, and within 2.2% for a spring or autumn week. When integrated with a building model, the results show 21% cost savings on energy in the mid-Atlantic region, and 27% savings in the Great Lakes region. Moreover, energy cost analysis is conducted to investigate the economic effect of CHP systems.

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1. Introduction

Residential energy consumption represents 22% of the total energy consumption in the United States. This includes energy for heating and cooling, humidity control, lighting, and residential appliances. When combined with commercial buildings, 41% of all energy consumption in the United States is from buildings [1]. Accordingly it is appropriate to search for energy saving measures and efficiency improvements. One possibility for improvements is the use of small-scale CHP (combined heating and power), called micro CHP. The CHP is defined in the context of this work as the combined generation of electricity and heat from a single fuel source. This most often refers to the use of an engine to generate electricity and the engine waste heat for space and/or water heating applications.

A micro CHP has been the topic of considerable research thanks to its potential savings, particularly in a more distributed grid-style energy system. Research focusing on modeling the micro CHP has largely been aimed at determining ideal applications for the CHP and evaluating the financial feasibilities of such applications. The International Energy Agency published, as part of a series of

reports, the Annex 42 report in which several fuel cell and combustion powered CHP devices are modeled and tested [2]. Also note-worthy is a paper by Hawkes and Leach (2005) that shows the effects of temporal precision in modeling on the results of simulation. They demonstrated that using typical 1-h-time step load profiles resulted in significant underestimation of the required system capacity [3]. Kelly et al. (2008) investigated about a 0.75 kW Stirling cycle CHP unit and a 5 kW IC (internal combustion) engine. They also modeled a building and ran an assortment of one-week simulations to evaluate CHP performance in a residential application [4]. Dentice d'Accadia et al. established a test facility to test small-scale CHP systems, and compiled a summary of state-of-the-art technologies including fuel cell, IC and Stirling engine systems [5]. Entchev et al. (2004) developed two identical houses with which to test CHP-aided homes against a baseline, with a 0.74 kW electrical output Stirling engine system [6]. The same facility, in a report by Bell et al. (2003) was used for one- to two-day tests of the Stirling system, showing total system efficiencies of approximately 82% in spring-time operation, and the small Stirling unit generating 25–43% of the total daily electricity consumption at the house [7]. De Paepe et al. (2005) compared various CHP systems to a reference house model to compare energy savings. Their findings showed an existing Stirling engine to consume as little as 73% of the energy consumed in the reference case. Their study also showed financial feasibility of all the CHP systems modeled, particularly with the

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Nomenclature		RPM	revolutions per minute
		SEER	seasonal energy efficiency ratio
		TMY	typical meteorological year
<i>Symbols</i>			
A	heat transfer area of heat exchanger	<i>Subscript</i>	
c_p	specific heat	c	cold-side
F	waste heat fraction	ci	cold-side inlet
$\dot{m}$	mass flow rate	co	cold-side outlet
Q	heat or power	$comp$	component
$\dot{Q}$	heat transfer rate or power	e	electrical
T	temperature	$elec$	electrics
U	overall heat transfer coefficient	f	fuel
ε	effectiveness	h	hot-side
η	efficiency	hi	hot-side inlet
<i>Acronyms</i>		ho	hot-side outlet
CCHP	combined cooling, heating and power	in	inlet
CHP	combined heating and power	min	minimum
EGR	exhaust gas recuperator	max	maximum
HVAC	heating, ventilation and air-conditioning	out	outlet
IC	internal combustion	p	primary energy
PHX	plate heat exchanger	sh	space heating
PID	proportional-integral-derivative	T	total
PLR	part load ratio	th	thermal
PM	prime mover	w	water

ability to over-produce and “sell back” to the grid [8]. Another overview of CHP, buildings, and specifically residential loads was performed by Dorer and Weber (2009), including system modeling with Matlab [9]. They also used TRNSYS [10] to evaluate performance with whole-building models. Their results clearly show that different CHP systems are advantageous in different buildings. Ren and Gao (2010) investigated two typical micro CHP alternatives which are using gas engine and fuel cell for residential buildings. They concluded that the fuel cell system is recognized as a better option for the examined residential building from both economic and environmental points of view [11].

Various configurations of domestic CHP systems have been evaluated for different thermal loads. Wu and Wang (2006) presented an overview of CCHP (combined cooling, heating and power), detailing the use of waste heat-powered cooling systems such as absorption chillers over a spectrum of size ranges [12]. Few et al. (1997) modeled the domestic CHP system using a heat pump [13]. Jalalzadeh-Azar (2004) showed the advantage of thermal load following control strategies as opposed to grid-isolated, electrical load following systems [14]. Peacock and Newborough (2007) investigated using aggregate load control as opposed to the typical heat load following method [15]. Cardona and Piacentino (2003) studied about a methodology for sizing a trigeneration plant in Mediterranean areas. The optimum size was decided based on the aggregate thermal demand [16]. Lund and Munster (2003) evaluated grid-level control strategies for CHP- and wind-heavy energy systems [17]. In addition, Huowing et al. (2008) evaluated the effect of demand and economic uncertainties with regards to such systems [18]. Cost minimization for a local utility was evaluated by Henning (1998), demonstrating impact of price and possible incentive programs on the short-term viability of CHP [19]. Using the larger-scale reference of a hotel or hospital, Cardona et al. (2006) evaluated the balance between economic and energetic or environmental savings in the case of CHP-heat pump hybrids. This work illustrates the existence of competing goals, as the work suggests that the manager seeking optimal financial benefit will realize far less environmental or energetic savings than that are

possible [20]. TeymouriHamzehkolaei and Sattari (2011) investigated technical and economic feasibility of using micro CHP in the different climate zones both economic and environmental points of view [21]. Sanaye and Ardali (2009) showed the payback period estimate for micro-turbine CHP systems ranging from 30 to 350 kW. Other than IC engines, systems including Stirling engines and fuel cells were being considered [22]. Kong et al. (2003) evaluated the efficiency and feasibility of CCHP systems driven by Stirling engines and using absorption chillers [23].

It was found out from literature review that there were very limited studies about modeling of CHP systems validated with high-resolution data in both steady state and transient test conditions, which is related to deviation between simulation and experimental results. Moreover, when this CHP system was applied for the building applications, the effect of the regional variation on performance was not investigated. In our study, modeling of the micro-CHP system is validated with high-resolution experimental results. Then a validated micro-CHP system model is used to evaluate its performance when integrated with a building model under different weather conditions. Then, energy cost analysis is conducted to investigate the economic benefit of CHP systems.

2. Approach

2.1. Laboratory configuration

The PM (prime mover) used was a 272cc single cylinder Otto cycle engine running on natural gas with a 4.0 kW of the electric capacity. It was manufactured by Marathon Ecopower (model number: 336016). The electrical generator was a shaft-mounted, water cooled permanent magnet generator, coupled directly to the engine. The generator output was inverted to the desired frequency and voltage by a single phase inverter. The engine speed was regulated by modulating the shaft torque imposed by the generator, allowing the engine to operate at open throttle across all part loads. As shown in Fig. 1, the engine was water cooled, and waste heat was recovered from the engine cooling jackets, oil

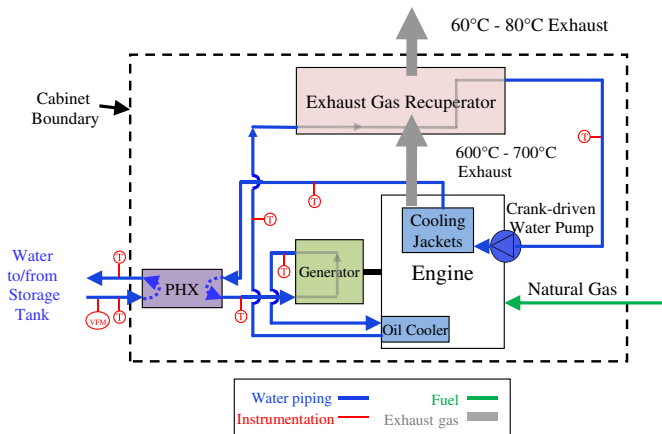


Fig. 1. Schematic diagram of PM (prime mover) in CHP system.

cooler, EGR (exhaust gas recuperator), and the generator. These heat recovery units are connected in series. Approximately 10–20% of waste heat was lost, depending on engine speed. The dashed line in Fig. 1 represents the boundary of the PM. The cooling sequence and approximate percentage of total waste heat recovery at full load is

PHX (Plate heat exchanger) → Generator (2%) → Oil cooler (8%) → EGR (44%) → Cooling jackets (35%) → PHX

Fig. 2 shows the overall laboratory configuration of the CHP system investigated. The main purpose of the PM's heat recovery components is to provide useful heating, which can be stored in the 832.8 L buffer tank or diverted to a radiator. The storage tank was 1.77 m high and approximately 0.77 m in diameter, with a rounded top and bottom. It was made of steel and covered with 0.0254 m thick flexible polyurethane foam insulation. Water lines around the water tank were connected as follows. The inlet from the PM supply line and the outlets for space heating and domestic hot water were located at the same level as the top in-stream thermocouple. The outlet to the PM return line and the inlet of the water tank for tap water and space heating return line were located at the same height as the lowest in-stream thermocouple.

2.2. PM model

The IC engine was modeled using a component in TRNSYS called Type 907. This component uses a table of empirical performance

Table 1
Parameters for IC engine components.

Parameter units	Unit	Value
Maximum power output	kW	3.993
Number of intake temperatures	Each	2
Number of part load ratio points	Each	6
Specific heat of jacket water fluid	$\text{kJ kg}^{-1} \text{K}^{-1}$	4.19
Specific heat of oil cooler fluid	$\text{kJ kg}^{-1} \text{K}^{-1}$	4.19
Specific heat of exhaust air	$\text{kJ kg}^{-1} \text{K}^{-1}$	1.007
Specific heat of generator fluid	$\text{kJ kg}^{-1} \text{K}^{-1}$	4.19
Rated exhaust air flow rate	kg h^{-1}	3.83

data to determine the operating outputs for an IC engine given set input conditions. The parameters used for calculations and their default settings are shown in Table 1. The component requires inputs including the desired electrical output, the ambient temperature, and the cooling fluid mass flow rate. The inputs are read at the beginning of each calculation iteration from other components. The component by default has a post-cooler. However, the fluid properties of water are input for this component, and it simply acts as the generator heat recovery component, and is referred to as such hereafter. The fuel consumption is calculated in the PM component based on the electrical efficiency as part of the performance map. It is calculated as a required heat input as shown in Eq. (1).

$$\dot{Q}_f = \frac{\dot{Q}_{\text{elec}}}{\eta} \quad (1)$$

where $\dot{Q}_f$ is the fuel input in kW, $\dot{Q}_{\text{elec}}$ is the electrical output of the PM in kW, and η is the system efficiency. In this paper, the electrical efficiency is considered to be the total efficiency of the engine at converting fuel heat to electrical output. The main input to the engine component is the desired electrical output of the engine. This can be provided as an input by the user or from another component, such as the control strategy components used in the transient simulations of this study. The desired output is converted to a PLR (part load ratio) and used to refer to a performance map which contains information on efficiency, exhaust flow and heat distribution. From this performance map, the fuel use and thermal output can be derived. The mass flow of the engine cooling fluid is also calculated as a function of PLR, as described in the [Steady state calibration and verification](#) section. The cooling water mass flow rate and the heat rejection to each component as found in the performance map are used to determine the temperatures in the cooling water loop using Eq. (2).

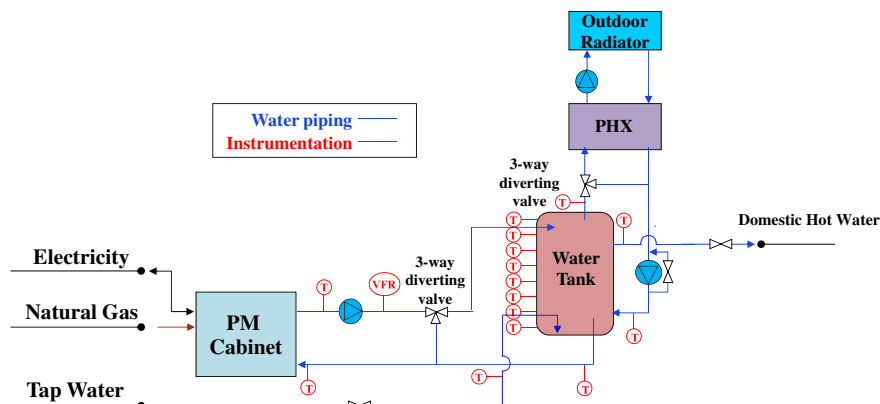


Fig. 2. Schematic of laboratory CHP system configuration.

$$T_{\text{out}} = T_{\text{in}} + \frac{\dot{Q}_{\text{comp}}}{\dot{m} \cdot c_p} \quad (2)$$

where $\dot{Q}_{\text{comp}}$ is the heat addition to the fluid stream from the component, $\dot{m}$ is the mass flow rate and c_p is the specific heat of the fluid. T_{in} and T_{out} are the temperature of the fluid flow into and out of the component, respectively.

2.3. Heat exchanger models

There are two important heat exchanger models in the PM model. An EGR, modeled as a Type 5b counter-flow heat exchanger, is used to cool the exhaust gas while adding heat to the cooling water flow. A flat-PHX also modeled as a Type 5b counter-flow heat exchanger acts as the interface between the engine cooling loop and the external water loop. The mathematical model for the heat exchangers is the same. The equation used is shown in Eq. (5)

$$\varepsilon = \frac{1 - \exp\left(-\frac{UA}{\dot{m}_{\min} c_{p_{\min}}} \left(1 - \frac{\dot{m}_{\min} c_{p_{\min}}}{\dot{m}_{\max} c_{p_{\max}}}\right)\right)}{1 - \left(\frac{\dot{m}_{\min} c_{p_{\min}}}{\dot{m}_{\max} c_{p_{\max}}}\right) \exp\left(-\frac{UA}{\dot{m}_{\min} c_{p_{\min}}} \left(1 - \frac{\dot{m}_{\min} c_{p_{\min}}}{\dot{m}_{\max} c_{p_{\max}}}\right)\right)} \quad (3)$$

where ε is the effectiveness, $c_{p_{\min}}$ and $c_{p_{\max}}$ are the minimum and maximum of the specific heats of the working fluids, U is the overall heat transfer coefficient, and A is the heat transfer area of the heat exchanger. $\dot{m}_{\max} c$ and $\dot{m}_{\min} c$ are the flow rates corresponding to the fluid with the maximum and minimum specific heat. The value of ε is used to determine the outputs of the heat exchanger using Eqs. (4) and (5).

$$T_{\text{ho}} = T_{\text{hi}} - \varepsilon \left(\frac{\dot{m}_{\min} c_{p_{\min}}}{\dot{m}_h c_{ph}} \right) (T_{\text{hi}} - T_{\text{ci}}) \quad (4)$$

$$\dot{Q}_T = \varepsilon \cdot \dot{m}_{\min} c_{p_{\min}} (T_{\text{hi}} - T_{\text{ci}}) \quad (5)$$

where T_{ho} and T_{hi} are the hot-side inlet and outlet temperatures, respectively, $\dot{m}_h$ is the hot-side mass flow rate, T_{ci} is the cold-side inlet temperature, c_{ph} is the hot-side fluid specific heat, and $\dot{Q}_T$ is the total heat transfer rate across the heat exchanger.

In the case of the PHX for which the fluids on both sides of the heat exchanger are water, Eq. (3) is reduced to Eq. (6).

$$\varepsilon = \frac{1 - \exp\left(-\frac{UA}{\dot{m}_{\min} c_{p_{\min}}} \left(1 - \frac{\dot{m}_{\min} c}{\dot{m}_{\max} c}\right)\right)}{1 - \left(\frac{\dot{m}_{\min} c}{\dot{m}_{\max} c}\right) \exp\left(-\frac{UA}{\dot{m}_{\min} c_{p_{\min}}} \left(1 - \frac{\dot{m}_{\min} c}{\dot{m}_{\max} c}\right)\right)} \quad (6)$$

The specific heats of the water and exhaust gas are parameters in TRNSYS, meaning they do not vary over time. Therefore, empirical approximations were used.

2.4. Tank model

In transient operation and building application testing, a storage tank was modeled. The model is a fluid-filled, constant volume tank. It is divided into isothermal temperature nodes to model stratification. The number of nodes affects the stratification of the tank, with more nodes leading to a greater stratification. For the purposes of this simulation, 39 nodes (numbered 1 at the top through 39 at the bottom) were used. This value produced a balance between accuracy and simulation time. The coefficient of

thermal loss from the tank to ambient was determined through empirical data of the tank, at the fully heated state. The value of $3 \text{ W m}^{-2} \text{ K}^{-1}$ corresponds to an R value of 0.33 or the equivalent of approximately 0.0127 m thick high density fiberglass insulation was used for tank insulation.

2.5. Simulation time step

The experimental data acquisition time step was 10 s. For the sake of simulation speed, the simulation time step used was 20 s for steady state and transient testing. The effect of this on the simulation outputs was negligible. However, the simulation time was significantly shorter than using the 10 s data acquisition time step as the simulation time step. For the building model integration, the initial time step was selected as 5 min for simulation time.

3. Steady state test configuration

The steady state model configuration was simplified for the purpose of calibrating the PM. This was the condition in which the supply and return lines bypass the storage tank and flow directly to the heat exchanger to the radiator loop. In the experimental configuration the PM outlet was pumped to a diverter, which was controlled by a PID (proportional-integral-derivative) algorithm to maintain the appropriate return temperature to the PM which is set to be 60°C . The diverter sent hot water to the return, as well as to the heat exchanger and outdoor radiator, where heat was dumped to simulate space heating. For calibration purposes, the heat exchanger and radiator were not significant, as the return temperature was set by the PID controller, and the return flow rate was constant. Therefore, the heat exchanger and outdoor radiator were not modeled here.

3.1. Steady state calibration and verification

The steady state calibration was accomplished by using laboratory experiments in which the PM was allowed to operate at a fixed RPM (revolutions per minute) for an extended period while dumping heat to the outdoor radiator. The laboratory measurements were used as inputs where applicable to control the laboratory model, and the model itself was tuned to match the simulated outputs to the results recorded in the laboratory. In steady state calibration, the return temperature and flow rate on the water-loop side of the PM was fixed. In addition, the temperature of the cabinet air inlet to the engine was used as the exhaust gas inlet temperature for the PM. The engine cooling water mass flow rate was derived as a linear fit to laboratory readings for each RPM. The equation derived for this mass flow rate, in kg s^{-1} , is shown in Eq. (7).

$$\dot{m} = 0.2629 \cdot \text{PLR} + 0.0441 \quad (7)$$

Laboratory results were used to derive the engine efficiency, total waste heat and the fraction of waste heat to each heat recovery component. The efficiency is calculated as Eq. (1). The heat recovered by each component in the laboratory is calculated as Eq. (8)

$$\dot{Q}_{\text{comp}} = F_{\text{comp}} \cdot \left(\left(\frac{\dot{Q}_{\text{elec}}}{\eta} \right) - \dot{Q}_{\text{elec}} \right) \quad (8)$$

where F_{comp} is the fraction of waste heat to that component obtained from the performance map. The fraction of waste heat to the environment is calculated by Eq. (9).

$$F_{\text{env}} = 1 - \sum F_{\text{comp}} \quad (9)$$

The heat recovered by each component was used to determine the temperature of the fluid exiting in that component using the relation in Eq. (10).

$$\dot{Q}_{\text{comp}} = c_p \dot{m}_{\text{engine}} (T_{\text{comp,out}} - T_{\text{comp,in}}) \quad (10)$$

where $\dot{m}_{\text{engine}}$ is the water mass flow rate to the engine. $T_{\text{comp,out}}$ and $T_{\text{comp,in}}$ are the water outlet and inlet temperature of each component.

By tuning the parameters for η and F_{comp} the heat recovery components except for the EGR were calibrated. The outputs were averaged over the time in which the system was in steady state operation. By using Eqs. (8)–(10), the efficiency and waste heat fractions were determined.

The EGR and PHX were tuned in a similar manner to the heat recovery components, except the UA -value was tuned rather than the F_{comp} value.

3.2. Steady state test results

The steady state test and simulation results are shown in Fig. 3. It shows the outputs of the steady state simulation in comparison with the laboratory outputs for the same conditions. The model was given as inputs from the laboratory; the engine intake temperature, the return water temperature and flow rate, and the engine operating set point. The maximum error in thermal production was 1.7%, well within the experimental measurement uncertainty of approximately 3.5%. The component with the greatest discrepancy between experimental and simulation output was the EGR. This error ranges from 0.2 to 3.3% of the total EGR heat recovery. The primary cause of this error is the specific heat approximation used for the exhaust gas. As discussed above, the specific heat of the exhaust gas had to be approximated as a parameter using average gas conditions through the PM system. This approximation resulted in a slight underestimation of the heat recovered from the exhaust gas at high RPM levels. As a consequence, the difference in thermal output of the PM coefficient varied with RPM. However, on a system level the thermal output error was at maximum 1.7% and therefore, the results might be considered satisfactory.

4. Transient test configuration

Transient testing was conducted in two configurations: first to validate the PM without a tank model, and second with a tank model in place to evaluate the performance of the tank as it would be seen in a real-world application. In addition, the tank was tested alone to determine the nodes in the model tank which best

matched the temperature profiles of the thermocouples in the laboratory tank. The PM control strategy was not published by the manufacturer. Therefore, the control strategy had to be derived from experimental results before any transient tests could be performed. This was done by observing the actual tendencies of the laboratory PM in transient testing. The PM had some consistent set points which could be derived and applied to the model. Some note-worthy control points include:

- After turned off, the PM turns back on when the tank middle temperature reaches 30 °C, and immediately starts its full-load operation.
- The PM remains in full-load operation until the lower temperature reaches 50 °C.
- The PM reduces its output in proportion to the temperature change in the tank until the PLR of 0.39, at which point it may remain until the tank is fully charged.
- The PM turns off when the lower temperature reaches 70 °C.

The first stage of transient testing was validation using the PM and control strategy only with experimental results as inputs. The tank top, middle and bottom temperatures from experimental results were fed back to the control strategy. In addition, the PM return temperature was fixed to the laboratory output. The model was tested first using the engine inlet air from the experimental results and then using a modeled inlet air temperature. For the second stage, as is discussed later, the tank model was implemented to evaluate the difference it caused.

The response of the cabinet air temperature to engine operation was modeled using the Type 88 single-zone lumped capacitance building model of TRNSYS. This component uses internal gains, ventilation and infiltration to calculate the space conditions of a simple zone using edge loss coefficients and zone capacitance. The dimensions were taken from manufacturer data for the PM cabinet. Since the actual air infiltration rate was not known, an approximation was used in order to achieve appropriate engine inlet temperatures. The infiltration of ambient air was set to equal to the mass flow rate of exhaust out of the cabinet, and the capacitance of the cabinet was tuned empirically. In order to minimize computation time the temperature outside the cabinet was entered as an average value rather than read from laboratory data. The effect of these approximations was minimal. The transient model also requires additional components to those already discussed to simulate the water loop between the PM and tank. A pump, modeled with component Type 3d, circulates water from the PM heat exchanger to the tank. The pump had a flow rate of 0.273 kg s⁻¹, the average in the laboratory. The heat loss of the pump to the fluid stream was neglected, as the measurement uncertainty of the laboratory thermocouples before and after the pump prevented any accurate value from being applied. The final configuration of transient testing was with the tank model in place. The ambient temperature, tap water temperature, and space heating and domestic hot water loads were the only external inputs to the model.

4.1. Load profiles

Two load profiles were applied to the system for transient validation and testing. The first was a shoulder week, designed to simulate a typical spring or autumn month. This load profile included both space heating and domestic hot water consumption. The second load profile was a typical summer week, which included only a domestic hot water load profile. The domestic hot water load profiles were acquired from the International Energy Agency and National Resources Canada [2]. The domestic hot water load was based on an assumption of approximately 200 L per day of

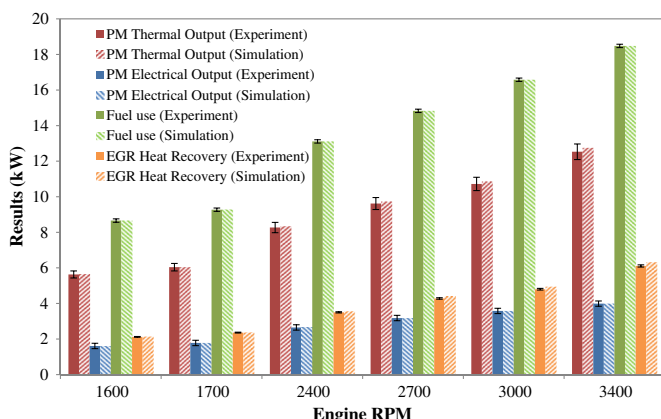


Fig. 3. Steady state test results.

hot water consumption, which is typical of a three-person household [2]. It used 30 s time steps, meaning that each domestic hot water load was sustained evenly for 30 s. The load profile had a maximum flow rate of 0.091 kg s^{-1} , or approximately 5.5 L min^{-1} . The space heating load was derived from the building model simulations for the mid-Atlantic region [24]. The maximum space heating load was 15.1 kW. The average load for the week was 5.12 kW, and the total space heating required was 860.4 kWh. In order to impose the space heating load on the tank, it is necessary to calculate a flow rate for a given load. For this purpose, the following equation was used to control the heating water flow rate using Eq. (11):

$$\dot{m}_{\text{sh}} = \frac{\dot{Q}_{\text{sh}}}{c_p \cdot (T_{\text{w,out}} - T_{\text{set}})} \quad (11)$$

where $\dot{m}_{\text{sh}}$ is the space heating water flow rate, $\dot{Q}_{\text{sh}}$ is the load from the load profile, c_p is specific heat, and $T_{\text{w,out}}$ is the temperature of water exiting the tank for space heating. 25.5°C was used as a reference temperature, T_{set} as it is the average return temperature to the tank from the space heating configuration, as controlled in the laboratory.

4.2. Transient test results

Transient testing was performed in four variations. The first two were validation tests, in which the PM component and control strategy were given the tank temperatures and return water temperature from experimental results as inputs. This was done for both the shoulder week and summer load profiles. The second element of testing was done with a tank model in place. This test was done to show the difference in performance between the validated PM model alone and the PM model with a non-validated tank model. A TRNSYS output for the shoulder week simulation is shown in Fig. 4. As is evident in the figure, the state of the PM does not match until approximately 10 h of simulation time has passed. The reason for this discrepancy is that the control strategy of the PM is dependent upon the previous state of the PM itself. As the figure shows, the data set used in this simulation begins with the PM already operating at part load. Since the simulation did not include the PM history before time equals zero, it did not respond in the same way to the load profile. However, by 10 h the simulation and experimental results have reached equivalent states. Another discrepancy occurs in this simulation at approximately 105 h simulation time. In this instance, the PM in the laboratory turned off slightly earlier than was typical. In order to assess the PM outputs and control strategy, instances of known and unavoidable

anomalies were neglected for validation. Finally, the laboratory test stopped slightly short for the week.

Fig. 5 shows an individual instance of the PM operating. As can be seen in the figure, there was a strong qualitative agreement between the operating status of the model and laboratory PMs. In the tune-down period beginning at approximately 81 h, the difference in the decision frequency of the model and laboratory PM can be seen. The model made PLR adjustments every time step, in this case every 20 s. The PM in the laboratory made PLR adjustments approximately in every 15 min. Therefore, the experimental PM adjusted in steps while the model PM transited more smoothly. It should be noted that the simulation time step was selected to represent the experimental results most closely. The engine operated primarily in two states: full-load operation and a low shoulder state of on average 1.48 kW, or a PLR of 0.37. This low shoulder state can be seen between approximately 82 and 84 h in Fig. 5. This is matched in the model by use of sticking functions, to prevent the model PM from frequently varying into and out of the full-load or low shoulder condition. The model was also run with the tank simulated. In this case, the input to the model was the load profile and ambient air conditions. The results of this simulation are shown in Fig. 6. It can be observed that there were more discrepancies than when the tank model was not simulated while the model and laboratory PMs behaved similarly. An example of this is the dip in PM output that occurred at hour 23.

Fig. 7(a) shows the variation of tank temperatures and PM output for 20–40 h of the shoulder week test. Fig. 7(b) shows the tank temperatures and PM output for the simulation during 20–40 h of the same test. The tank temperatures correspond to the left axis, while the PM outputs correspond to the right axis. At approximately hour 23, the model tank's lower temperature crosses 50°C , initiating the ramp-down stage of the PM control strategy and causing a slight dip in output. This dip did not occur in the laboratory test. This discrepancy was caused in the stratification difference in the tank models. As can be seen in the figure, the simulated tank bottom temperature was higher than the corresponding laboratory temperature, and increased more gradually.

Fig. 8 shows a similar TRNSYS output for the summer week simulation. As this load was dominated by standby losses, the PM was not required to operate as frequently or for as great a duration. Since the periods for which the PM operated were short, discrepancies in which the model or laboratory engine ran longer would have a comparatively large percent error. During the simulation period of 33–36 h, the model overestimated the energy outputs by 13–18%. During simulation period of 142–146 h, however, the simulation underestimated the energy outputs by 10–15%. One

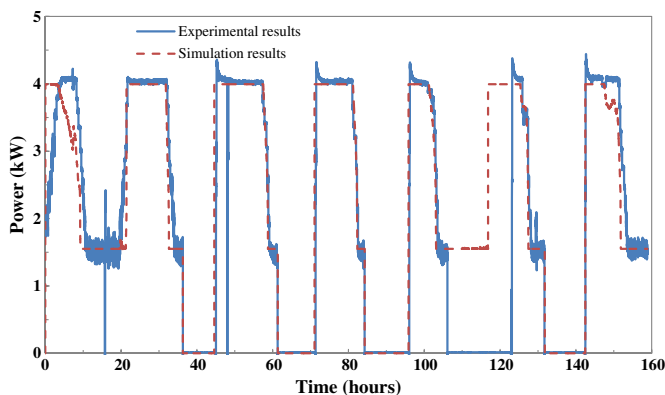


Fig. 4. Variation of power output for shoulder week without tank model.

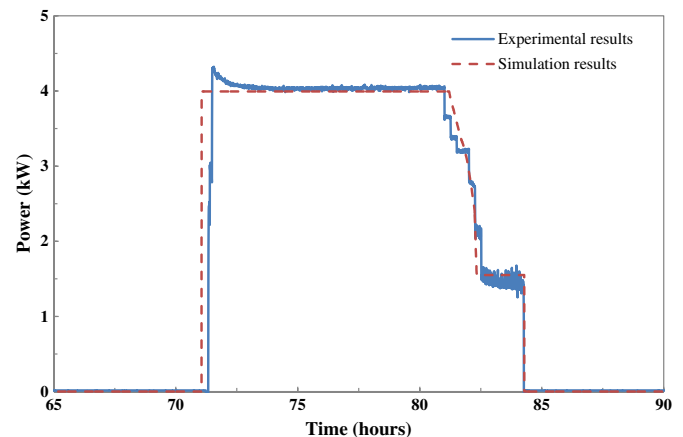


Fig. 5. Variation of power for shoulder week during single PM cycle.

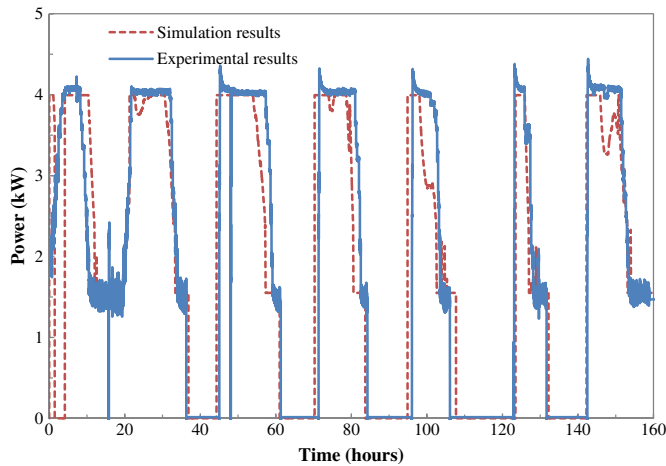


Fig. 6. Variation of power for shoulder week with tank model. (a) Experimental results (b) Simulation results.

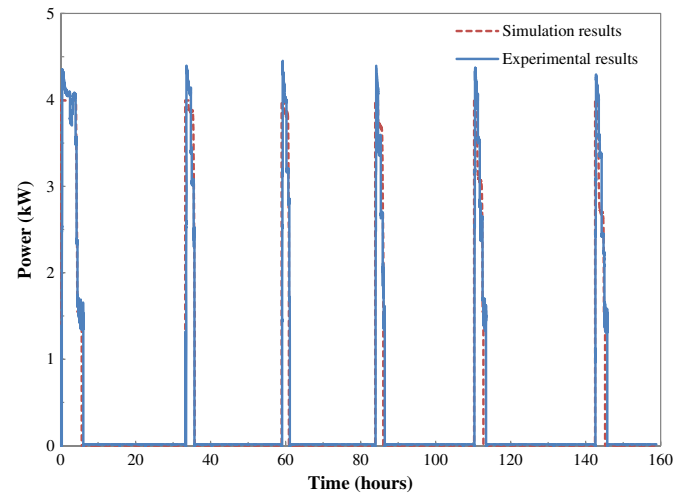


Fig. 8. Variation of power output for summer week without tank model.

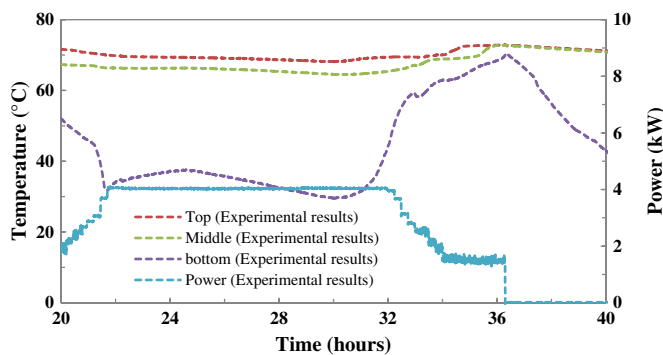
such instance is illustrated in detail by Fig. 9. As this example demonstrates, the discrepancies over individual operating periods were difficult to overcome. For the accumulated outputs of the whole week, these errors balanced over the course of the entire week-long simulation. This suggests that the cause of these differences rests in normal variations in operating conditions and measurement accuracy. Therefore, the model could be considered valid for this test.

As with the shoulder week simulation, the summer week was also simulated with the tank model incorporated. Fig. 10 shows the PM outputs from the simulation and the experiment. The simulation PM operated more frequently, but for shorter durations than that of the experimental PM. Since standby losses were the

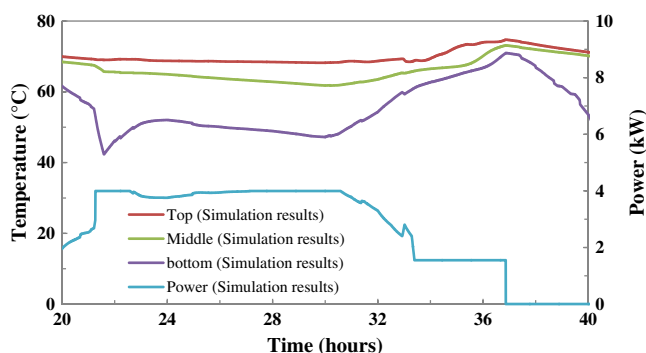
dominant factor in determining when the PM operated, rather than a scheduled load as in the shoulder week test, it was to be expected that there would be some discrepancy.

Tables 2 and 3 show the summarized results of transient testing with and without the tank model incorporated. Each of these tables shows the simulation and laboratory fuel input, electrical output, and thermal output for the given time period. Table 2 shows the shoulder week, and Table 3 shows the summer week to be described in more detail as below. As can be seen in the tables, the CHP model was validated for transient testing without a tank, to within 2.2% for all energy outputs for the shoulder week testing and within 4.8% for all energy outputs for summer week testing. The spring test with the tank model was accurate to within 2.1%, while the summer test with the tank model was accurate to within 7.4%. The reason for the higher summer tank test error is that the summer test was dominated by standby losses. As the thermal losses of the tank could not be directly measured but rather only inferred, these losses were estimated. This estimate was performed to attain a near-zero error for an individual test. However, variations in laboratory conditions over the course of testing made it impossible to simulate with continued accuracy. Therefore, in a standby situation, some inaccuracy in the model is expected.

From the above tables it can be seen that the engine performed much better when operating more frequently, as in the spring tests. A comparison can be made using following equations:



(a) Experimental results



(b) Simulation results

Fig. 7. Variation of tank temperatures and PM output for shoulder week.

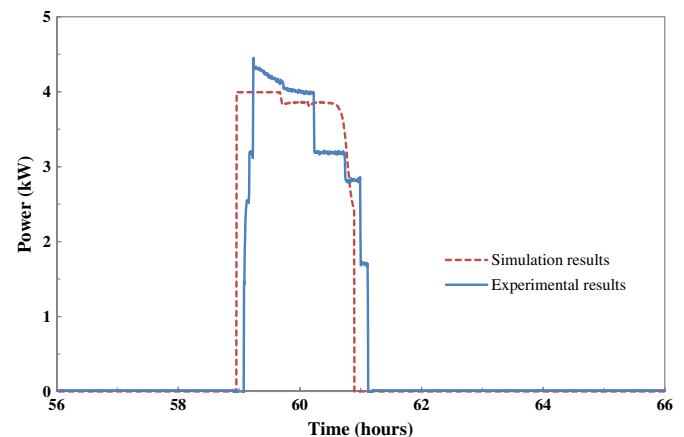


Fig. 9. Variation of power output for summer week during single PM cycle.

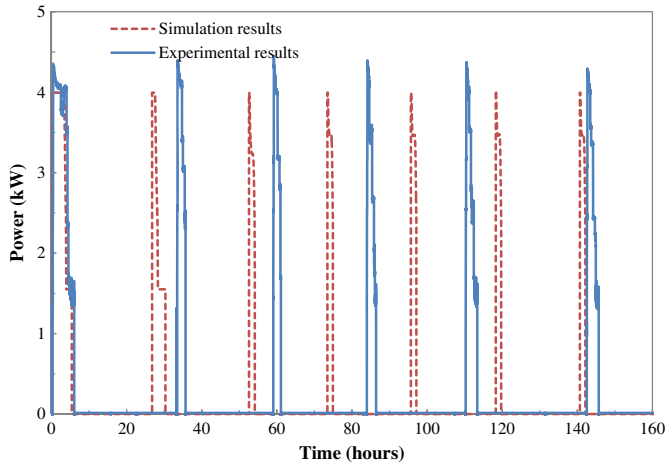


Fig. 10. Variation of power output for summer week with tank model.

$$\eta_e = \frac{\dot{Q}_e}{\dot{Q}_f} \quad (12)$$

$$\eta_t = \frac{\dot{Q}_{th}}{\dot{Q}_f} \quad (13)$$

$$\eta_{comb} = \frac{\dot{Q}_e + \dot{Q}_{th}}{\dot{Q}_f} \quad (14)$$

where η_e represents electrical efficiency, η_t represents thermal efficiency, η_{comb} represents combined efficiency, $\dot{Q}_e$ is the electrical power generated, $\dot{Q}_f$ is fuel used, and $\dot{Q}_{th}$ is thermal power generated. These efficiencies are tabulated for the entire test periods in Table 4.

Although the PM operated for a longer period during the heating season, Table 4 shows that the PM operated with a greater average electrical efficiency during the summer – when the operating time spent at full load was a larger percentage of total

operating time – than the spring, when long periods at the low shoulder RPM condition were more common. Correspondingly, more fuel used for heating during the spring season. The combined efficiency was slightly better in the spring test. This metric does not account for the difference between heat production and electrical generation in usefulness, storage or cost. As it will be discussed later in the CHP-building integration simulation section, the CHP performance was in fact far better during the winter.

5. CHP-building integration simulations

5.1. Building model

A building model was developed for the purpose of energy conservation measure. The building modeled was a 228.9 m². It was two stories, with four 28.6 m² rooms on the bottom floor, and two 57.2 m² rooms on the top floor. In addition, there was an unconditioned basement and attic. The building orientation was such that the long side of the building runs east–west. The conditioned building zones were modeled with 2.5 m ceilings, resulting in a total conditioned volume of 558 m³. The basement was 131.3 m² with a 2.1 m high ceiling. The basement was assumed to be surrounded by ground on all sides. The attic was of the same square footage as the basement, but with a pitched ceiling reaching a height of 3.3 m. The roof surface area was 128 m². The building was modeled with 15% of each external wall area taken up by windows. The windows used did not include any special shading or glazing. The building walls were modeled, 0.4 m on center framing, with gypsum drywall on both sides of the interior walls. The exterior walls had brick on the outdoor side, and included fiberglass batt cavity insulation. The wall had a thermal resistance of $R = 12.5$. The basement walls and floor were concrete and had a thermal resistance of $R = 10$. In addition, there was $R = 17$ insulation between the basement and first floor, and $R = 33$ between the top floor and the attic. The infiltration rate was 0.474 air changes per hour.

Weather conditions were applied to the building via TMY2 (TMY (typical meteorological year) data) [25]. This data is a collection of the “most typical” of each month over a 30-year collection period. The entire month was used in order to maintain the stochastic nature of real weather patterns, while avoiding major

Table 2
Transient shoulder week results with and without tank model.

Property	No tank model			Tank model		
	Shoulder week, 10–105 h			Shoulder week, 10–135 h		
	Fuel input (kWh)	Electric output (kWh)	Thermal output (kWh)	Fuel input (kWh)	Electric output (kWh)	Thermal output (kWh)
Simulation results	987.9	206.4	660.7	1094	226.5	720.6
Experimental results	993.6	207.3	646.1	1110	231.3	718.6
Absolute difference	−5.7	−0.9	14.6	−16.3	−4.8	2
Percent difference (base 100)	−0.6	−0.4	2.2	−1.5	−2.1	0.3

Table 3
Transient summer week results with and without tank model.

Property	No tank model			Tank model		
	Summer week, 10–170 h			Summer week, 10–170 h		
	Fuel input (kWh)	Electric output (kWh)	Thermal output (kWh)	Fuel input (kWh)	Electric output (kWh)	Thermal output (kWh)
Simulation results	281.4	59.8	182.2	263.2	55.5	166.2
Experimental results	282.7	58.7	173.5	282.7	58.7	173.5
Absolute difference	−1.3	1.1	8.7	−19.5	−3.2	−7.3
Percent difference (base 100)	−0.5	1.8	4.8	−7.4	−5.8	−4.4

Table 4

Electrical, thermal and combined efficiencies over test period.

	Transient spring		Transient summer	
	No tank model	Tank model	No tank model	Tank model
η_e	20.9	20.7	21.3	21.1
η_t	66.9	65.9	64.7	63.1
η_{comb}	87.8	86.6	86.0	84.2

abnormalities. This means a hypothetical year could include the entire January of 1982, February 1990, March 1974, and so on. The building location was Sterling, VA as a baseline region (mid-Atlantic) and Madison, WI as a colder region (Great Lakes).

The HVAC (heating, ventilation and air-conditioning) system consisted of a gas-fired furnace, vapor-compression air conditioner, vapor-compression dehumidifier and a humidifier. The furnace was modeled as a commercially available 15 kW furnace with an efficiency of 0.79. The furnace included a humidifier which was a passive wick system. The air conditioner was a 7 kW vapor-compression system with a SEER (Seasonal Energy Efficiency Ratio) of 11.

The thermostat controls were regulated by a scheduling system. The heating season was defined as lasting from simulation hours of 0–3500 and 6000–8760, or approximately mid-September to mid-May. Outside of these times the heating system was by default set to off. The cooling season was defined to be between 2500 and 7000 h, or early April–late October. Outside of this time, cooling did not occur. There was a shoulder period in which heating and cooling could both occur, between 2500 and 3500 h as well as 6000 and 7000 h. In modeling the daily setback schedules of the residence, it was assumed that during the cooling season there was a setback of 25.5–30 °C, and a setback of 20.5–16.5 °C during the heating season based on occupancy.

The hot water system took tap water assumed to be at the temperature of ground water 1 m below surface level – and heated it. By default, the system was a 0.3 m³ tank with gas heat with a set point of 50 °C. It should be noted that this system was a smaller tank and a lower set point than those used in the CHP model, which might introduce a source of error with the CHP and tank system

replacing the default hot water system. The hot water and space heating systems were replaced by the CHP with water tank configuration in order to evaluate the performance of the CHP system in this building model. The CHP system and water tank were installed exactly as described in the transient testing configuration. For space heating, a pump, fan and heating coil were modeled. The heating was assumed to be done with one central heating coil, using the air distribution system in place from the building model simulation. The space heating pump was given a flow rate of 0.139 kg s⁻¹, or 0.139 L s⁻¹. The power of this pump was 30 W. The heating fan was modeled initially as a single speed fan with a flow rate of 0.556 kg s⁻¹, equating to an air change rate of 0.261 air changes per hour. Its energy consumption was 745 W, within the typical range for a large residential, central blower. The fan forced air over a heating coil that used the bypass factor approach, which assumed that a fraction of the air stream reached temperature equilibrium with the heating fluid, while the remaining fraction was not affected. The two streams mixed to produce the outlet condition. The component was set to have a bypass fraction of 0.15, meaning that 15% of the air stream was unaffected by the heating coil. In addition, the unit was assumed to be off if either the water or air flow rate was zero.

In the baseline CHP simulation, net metering was assumed, meaning the utility paid the customer for electricity generation at a rate equal to what the customer paid for consumption. In 11 US states, net metering is required to be offered in some capacity. In some cases, there are restrictions, insurance requirements and additional equipment requirements which would affect the financials of net metering. However, no such costs were assumed in this simulation [26].

5.2. Building simulation results in mid-Atlantic region

Table 5 shows energy consumption of two building models: baseline (without CHP) and with CHP. In mid-Atlantic region, there is a 60% reduction in electrical consumption for a 50% increase in fuel consumption. The load in mid-Atlantic region is 32% electrical consumption if comparing kWh of gas consumption as well as electricity. The energy consumption of the building model in the

Table 5

Energy consumption of two building models in mid-Atlantic region.

Property	Baseline – No CHP			With CHP			
	Unit	Consumption	Energy	Property	Unit	Consumption	Energy
Space heating	kWh	15,973	Natural gas	CHP fuel	kWh	32,257	Natural gas
Hot water	kWh	5461	Natural gas	CHP net electric	kWh	–6070	Electric
Cooling	kWh	2411	Electric	Cooling	kWh	2452	Electric
Dehumidification	kWh	79	Electric	Dehumidification	kWh	75	Electric
Other loads	kWh	7499	Electric	Other loads	kWh	7499	Electric
Total	kWh	31,423	–	Total	kWh	36,213	–
Total electric	kWh	9989	Electric	Total electric	kWh	3956	Electric
Total gas	Therm	732	Natural gas	Total gas	Therm	1101	Natural gas

Table 6

Simplified annual energy cost of building models in mid-Atlantic region.

Property	Baseline – No CHP				With CHP				
	Unit	Value	Consumption (\$)	Cost (\$)	Property	Unit	Value	Consumption (\$)	Cost (\$)
Space heating	Therm	545.2	0.962	524.5	CHP fuel	kWh	1100.9	0.962	1059.1
Hot water	Therm	186.4	0.962	179.3	CHP net electric	kWh	–6070	0.128	–777.0
Cooling	kWh	2411	0.128	308.6	Cooling	kWh	2452	0.128	313.9
Dehumidification	kWh	79	0.128	10.1	Dehumidification	kWh	75	0.128	9.6
Other loads	kWh	7499	0.128	959.9	Other loads	kWh	7499	0.128	959.9
Total	–	–	–	1982.4	Total	–	–	–	1565.4

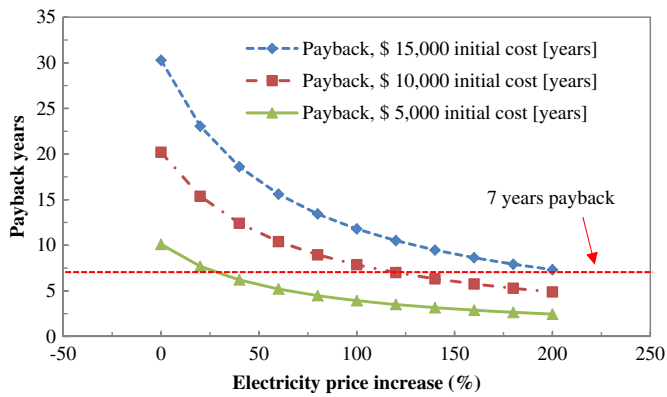


Fig. 11. Variation of payback years with electricity price in mid-Atlantic region.

baseline configuration is 31,423 kWh. Of this, 9989 kWh is electricity consumption and 21,434 kWh, or 732 therms, is natural gas.

Energy cost analysis is conducted to investigate the economic benefit of the CHP system. The average energy costs in the United States are applied to the simulation: natural gas price is \$0.962 and electricity price is \$0.128 [27]. Table 6 shows the simplified annual energy cost of building models without and with the CHP. The natural gas cost of the CHP system is higher than that of the baseline by 50%. However, the electricity is generated in the CHP system. When this electricity is used or sold back to grid, total energy can be saved from the CHP system by 21%.

Fig. 11 shows the variation of payback years with the electricity price. A simple payback period calculation assumes a PM price. Seven year payback is typically recommended for residential energy investments [28]. As the electricity price is increased, payback years reduced. When the initial cost of the CHP unit is \$5000 with the current electricity price, it takes 10.1 years of payback. This period increases up to 20.2 years for the initial cost of \$10,000, and 30.3 years for the initial cost of \$15,000. Lower initial cost could be possible with financial incentive which is offered to the owner, perhaps by the government or by the utility. When the electricity price is predicted to increase in the future, seven year payback can be achieved with the electricity price increase as following: 30% for \$5000, 120% for \$10,000, and 210% for \$15,000.

Table 7
Primary energy comparison of baseline building and building with CHP.

Property	No CHP	CHP	Difference (%)
On-site Gas [kWh]	21,434	32,257	−50.5
Electricity [kWh]	9989	3956	60.4
Primary Energy: 45% grid efficiency	43,631	41,048	5.9
Primary Energy: 40% grid efficiency	46,406	42,147	9.2
Primary Energy: 35% grid efficiency	49,973	43,559	12.8
Primary Energy: 30% grid efficiency	54,730	45,443	17.0

Table 8
Energy consumption of two building models in Great Lakes region.

Baseline – No CHP				With CHP			
Property	Unit	Consumption	Energy	Property	Unit	Consumption	Energy
Space heating	kWh	25,353	Natural gas	CHP fuel	kWh	42,081	Natural gas
Hot water	kWh	5543	Natural gas	CHP net electric	kWh	−7743	Electric
Cooling	kWh	1519	Electric	Cooling	kWh	1538	Electric
Dehumidification	kWh	47	Electric	Dehumidification	kWh	44	Electric
Other loads	kWh	7499	Electric	Other loads	kWh	7499	Electric
Total	kWh	39,961	—	Total	kWh	43,419	—
Total electric	kWh	9065	Electric	Total electric	kWh	1338	Electric
Total gas	Therm	1054	Natural gas	Total gas	Therm	1436	Natural gas

The savings of primary energy are also of interest for a CHP system. The primary energy savings are approximated in Table 7 for an assumed grid and transmission efficiency range of 30–45%. This calculation is simply based on Eq. (15).

$$\dot{Q}_p = \frac{\dot{Q}_e}{\eta} + \dot{Q}_f \quad (15)$$

where $\dot{Q}_p$ is total primary energy usage, $\dot{Q}_e$ is the total on-site electricity usage, η is grid and transmission efficiency, and $\dot{Q}_f$ is on-site fuel usage. Typical grid efficiencies range from 33 to 44%, with 7.2% transmission losses on average [29]. As can be seen in Table 7, the less efficient the grid is assumed to be, the better the CHP system performs in primary energy use, and therefore fuel consumption and emissions. These values are comparable to the results of a study by De Paepe et al. [8], who found savings for the same engine of 19% compared with a typical combined cycle power plant, 26% against an average fossil fuel plant, and 32% against the average power plant in Belgium [8]. A major difference between this study and the De Paepe et al. work is that they used results of a basic building simulation, without the CHP system, to predict when the system would operate and calculate savings, rather than simulate the transient operation of the engine. Therefore, their PM model may not accurately capture the control strategy of the real PM. Moreover, De Paepe et al. found a PM electrical output of 4.7 kW at maximum, and 25% efficiency, which was the advertised value from the manufacturer. Finally, the De Paepe et al. study baseline case had an approximately 10:1 ratio of heating primary energy use to electricity demand, compared with the approximately 2.1:1 ratio in this work.

5.3. Building simulation results in Great Lakes region

Since the PM performs best during a colder period, it is of interest to evaluate the performance of a CHP integrated building in a colder climate. The results of this simulation are compiled in Tables 8 and 9. As can be seen from Table 8, the CHP system electrical consumption of the Great Lakes region house is reduced by 85% for a 36% increase in fuel use as compared to the baseline. In mid-Atlantic region, there is a 60% reduction in electrical consumption for a 50% increase in fuel consumption. It stands to reason that in a climate in which the electrical consumption is a smaller portion of demand, a greater percentage change can be achieved when generating electricity on-site. The load in mid-Atlantic region is 32% electrical consumption if comparing kWh of gas consumption as well as electricity. In Great Lakes region, it is 23% electrical consumption. Compared with mid-Atlantic region, the Great Lakes region case uses 31% more fuel and generates 28% more electricity.

Similar comparisons can be drawn with costs, which are compiled in Table 9. In the base, net-metering case, the energy cost is reduced by 27% for the Great Lakes region simulation. In mid-Atlantic region, the savings is 21%. In Great Lakes region in the

Table 9
Simplified annual energy cost of building models in Great Lakes region.

Property	Baseline – No CHP				With CHP				
	Unit	Value	Consumption (\$)	Cost (\$)	Property	Unit	Value	Consumption (\$)	Cost (\$)
Space heating	Therm	865.3	0.962	832.4	CHP fuel	kWh	1436.2	0.962	1381.6
Hot water	Therm	189.2	0.962	182.0	CHP net electric	kWh	–7743	0.128	–991.1
Cooling	kWh	2411	0.128	308.6	Cooling	kWh	2452	0.128	313.9
Dehumidification	kWh	79	0.128	10.1	Dehumidification	kWh	75	0.128	9.6
Other loads	kWh	7499	0.128	959.9	Other loads	kWh	7499	0.128	959.9
Total	–	–	–	2293.0	Total	kWh	–	–	1673.9

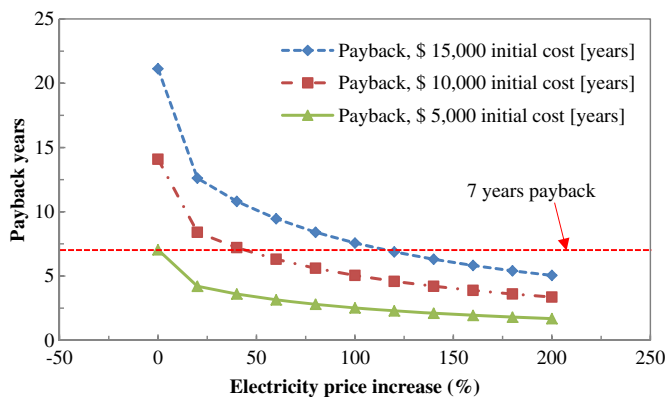


Fig. 12. Variation of payback years with electricity price in Great Lakes region.

baseline case, electricity consumption makes up 58% of total costs, while in Sterling it is 67%. These savings are reasonable. Therefore, a reduction in on-site electricity consumption has a greater impact on cost than a corresponding increase in on-site natural gas consumption.

Fig. 12 shows the variation of payback years with the electricity price. As the electricity price is increased, the payback years reduce. When the initial cost of the CHP unit is \$5000 with the current electricity price, it takes seven years for payback. These periods increase to 14 years for \$10,000, and 21 years for \$15,000 of the initial cost. As compared to the mid-Atlantic region, these periods are reduced by 30%. In order to have seven year payback period, the required electricity price increases are 50% for \$10,000, and 120% for \$15,000. These values are much shorter than those in mid-Atlantic area.

6. Conclusions

In this work, a micro-CHP model was developed and validated using high-resolution data acquired from experimental work. The model was tuned in steady state, and validated with experimental data for transient operation. The PM was found to have an electrical efficiency ranging from 19 to 22%, with an electrical output range of, on average, 1.6–4.0 kW and a thermal output range of 5.7–12.8 kW. The thermal output in the steady state simulation has an error of 1.7% at worst. In the simulation of one typical shoulder week, the thermal output error was 2.2%, the electrical output error was 0.5% and the fuel input error was 0.6%. With the tank model, the thermal output error was 0.3%, the electrical output was 2.2% and the fuel use error was 1.5%. In the simulation of one typical summer week, where standby losses and tank stratification have a stronger impact, the errors without the tank model were 4.8% for thermal output, 1.8% for electrical output, and 0.5% for fuel input. With the tank, the errors were 4.4% for thermal output, 5.8% for electrical output and 7.4% for fuel use.

Then a validated micro-CHP system model was used to evaluate its performance when integrated with a building model, using a single heating coil and providing domestic hot water. It was compared with a baseline residential HVAC and water heating case. The storage tank was used to provide domestic hot water and space heating, and the PM operated based on the same control strategy derived for transient simulations and validation. It can be observed from the results obtained here that in the mid-Atlantic region of the US, the micro-CHP system modeled is not financially viable at its current cost. The unit does save energy: the primary energy savings calculations show savings of 6–17%, depending on the assumed grid efficiency. In addition, the unit has 21% lower operating cost in the net-metering scenario. As the electricity price is increased, the payback period is predicted to reduce.

Then, a simulation is run in the climate of Great Lakes region, to demonstrate the importance of weather in CHP operation. Madison was selected because it is colder on average than the mid-Atlantic region, but sufficiently similar to allow the same building and HVAC model to be used for the simulation. The results of this simulation show the promise for the CHP in colder regions. The savings are greater in Madison, with a 27% cost savings, and the payback period of 7–21 years is substantially shorter, though still not investment-worthy.

From this work a number of general conclusions about the CHP can be drawn. First and perhaps most importantly, for the CHP to contend in the US energy market, costs need to come down significantly; a \$10,000 unit in Madison WI would have an approximately 14 year payback period, which begins to approach a marketable level if rebate or tax incentives exist. It is imperative to the viability of the CHP that some variation of net metering is offered, or else CHP units must be used in situations in which they rarely or never generate more electricity than is consumed. Purchasing battery storage, with CHP's cost being the most major inhibitor, would not be viable at this time. Also, it has been confirmed here that the CHP is of more value in cold climates. The savings in Madison are notably greater than those in Sterling. Lastly, it should be noted that all results obtained from this study were based on the Otto engine as the PM, so that it cannot be generalized for all the CHP applications.

Acknowledgments

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References

- [1] Annual energy review. U.S. Energy Information Administration, Office of Energy Statistics, U.S. Department of Energy; 2011.
- [2] Beausoleil-Morrison I. An experimental and simulation-based investigation of the performance of small-scale fuel cell and combustion-based cogeneration devices serving residential buildings. In Annex 42 of the International Energy Agency energy conservation 2008.

- [3] Hawkes A, Leach M. Impacts of temporal precision in optimization modeling of micro-combined heat and power. *Energy* 2005;30:1759–79.
- [4] Kelly NJ, Clarke JA, Ferguson A, Burt G. Developing and testing a generic micro-combined heat and power model for simulations of dwellings and highly distributed power systems. *Power and Energy* 2008;222:685–95.
- [5] Dentice D'Accadia M, Sasso M, Sibilio S, Vanoli L. Micro-combined heat and power in residential and light commercial applications. *Applied Thermal Engineering* 2003;1247–59.
- [6] Entchev E, Gusdorf J, Swinton M, Bell M, Szadkowski F, Kalbfleisch W, et al. Micro-generation technology assessment for housing technology. *Energy and Buildings* 2004;36:925–31.
- [7] Bell M, Swinton M, Entchev E, Gusdorf J, Kalbfleisch W, Marchand R, et al. Development of micro combined heat and power technology assessment capability at the Canadian centre for housing Technology. National Research Council; 2003. B6010.
- [8] De Paepe M, D'Herdt P, Mertens D. Micro-CHP systems for residential applications. *Energy Conversion and Management* 2005;47:3435–46.
- [9] Dorer V, Weber A. Energy and CO₂ emissions performance assessment of residential. *Energy Conversion and Management* 2009;50:648–57.
- [10] TRNSYS 17: mathematical reference, vol. 5. Madison, WI: Solar Energy Laboratory, University of Wisconsin; 2012.
- [11] Ren H, Gao W. Economic and environmental evaluation of micro CHP systems with different operating modes for residential buildings in Japan. *Energy and Buildings* 2010;42:853–61.
- [12] Wu DW, Wang RZ. Combined cooling, heating and power: a review. *Progress in Energy and Combustion Science* 2006;32:459–95.
- [13] Few PC, Smith MA, Twidell JW. Modeling of a combined heat and power plant incorporating a heat pump for domestic use. *Energy* 1997;22:651–9.
- [14] Jalalzadeh-Azar Ali A. A comparison of electrical and thermal load following CHP systems. *ASHRAE Transactions Research* 2004;110:85–94.
- [15] Peacock AD, Newborough M. Controlling micro-CHP systems to modulate electric load profiles. *Energy* 2007;32:1093–103.
- [16] Cardona E, Piacentino A. A methodology for sizing a trigeneration plant in Mediterranean areas. *Applied Thermal Engineering* 2003;23:1665–80.
- [17] Lund H, Munster E. Modeling of energy systems with a high percentage of CHP and wind power. *Renewable Energy* 2003;28:2179–93.
- [18] Huowang M, Ajah A, Heijnen P, Bouwmans I, Herder P. Uncertainties in the design and operation of distributed energy resources: the case of micro-CHP systems. *Energy* 2008;33:1518–36.
- [19] Henning D. Cost minimization for a local utility through CHP, heat storage and load management. *International Journal of Energy Research* 1998;22:691–713.
- [20] Cardona E, Piacentino A, Cardona F. Matching economical, energetic and environmental benefits: an analysis for hybrid CHCP-heat-pump Systems. *Energy Conversion and Management* 2006;47:3530–42.
- [21] TeymouriHamzehkolaei F, Sattari S. Technical and economic feasibility study of using micro CHP in the different climate zones of Iran. *Energy* 2011;36:4790–8.
- [22] Sanaye S, Ardali MR. Estimating the power and number of microturbines in small-scale combined heat and power systems. *Applied Energy* 2009;11:895–903.
- [23] Kong XQ, Wang RZ, Huang XH. Energy efficiency and economic feasibility of CCHP driven by Stirling engine. *Energy Conversion and Management* 2003;45:1433–42.
- [24] Mueller AC. Analyses of residential building energy systems through transient simulation. Master Thesis. College Park: University of Maryland; 2009.
- [25] TMY2, typical meteorological years, derived from the 1961–1990 National Solar Radiation Data Base. National Renewable Energy Laboratory; 1995.
- [26] Connecting to the grid project state interconnection standards for distributed generation. Interstate Renewable Energy Council; Jan 2010.
- [27] Average energy prices in United States: Mid-Atlantic Information Office. Available on the web, <<http://www.bls.gov/ro3/apwb.htm>>.
- [28] Staffell I, Green R, Kendall K. Cost targets for domestic fuel cell CHP. *Journal of Power Sources* 2008;181:339–49.
- [29] Technology options for the near and long term. United States Climate Change Technology Program; 2003.